Damping mechanisms for precision applications in UHV environment

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Surface analysis techniques based on a probe located with high precision such as the approaches with scanning tunneling microscopes (STM) and atomic force microscopes (AFM) have undergone significant advances and revolutionized the study of surfaces and nanometer size objects. For surface stability purposes, these experiments are often carried out in ultra high vacuum (UHV) conditions where the pressure ranges between 10^{-7} Pa and 10^{-9} Pa. The performance of these instruments is affected by external vibrations, so effective damping of the instrument is vital. In this article, we review some of the isolation methods such as active and passive damping and study the viability of these methods in a UHV environment. Finally, the application of some of these damping mechanisms for a scanning tunneling microscope (STM) that is currently being built and tested at the National Institute of Standards and Technology (NIST) is discussed.

KEYWORDS: Vibration damping, viscoelastic materials, eddy current damping

I. Introduction

Surface analysis techniques based on a probe located with high precision such as scanning tunneling microscopes (STM) and atomic force microscopes (AFM) have undergone significant advances and revolutionized the study of surfaces and nanometer size objects. For surface stability and achieving high resolution, these techniques are routinely used in ultra high vacuum (UHV) environment where the pressure inside the chamber varies between 10^{-7} Pa and 10^{-9} Pa. In addition, the instruments need to be isolated from external vibrations and as well damped. For example, to achieve atomic resolution using the STM, it must be provided with a vibration isolation system capable of reducing the external perturbations in order to obtain good stability in the tunneling junction. Vibrations that arise from people walking on the floor or the closing and opening doors are minimized by suspending the STM from the chamber using springs with eddy current dampers and by mounting the entire chamber on vibration isolation mechanism. Higher frequency vibrations that arise from mechanical pumps are suppressed with acoustical chambers and by using elastomer materials like Viton[I] for damping.

In this article, we review some of the methods that are used or potentially used for damping in UHV environments and discuss some of their advantages and disadvantages. This article is divided into six sections. Section II reviews basic definitions and classification of damping. Section III presents some traditional methods of passive damping such as viscous damping, magnetic damping, and damping using piezoelectric materials. Some of the techniques for isolating low and high frequency vibration are discussed in Section IV. Design case studies that incorporate some of these damping methods are described in Section V and details of the vibration isolation for UHV-STM currently being built and tested at NIST are described in Section VI.

II. Damping

Damping is the extraction of mechanical energy from a vibrating system, usually by conversion of energy into heat. It is broadly classified into two categories, passive and active damping. The principal difference between these two methods of damping is that in passive damping, the magnitude and direction of the force exerted by the damper depend only on the relative velocity across the damper. In active damping, an actuator is used to generate a force

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I *Certain commercial equipment, instruments, or materials are identified in this paper to foster understanding. Such identification does not imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that the materials or equipment identified are necessarily the best available for the purpose.*

opposing the motion, regardless of the relative velocity across it. An example of an active damping mechanism used in a vacuum system is described by Lan *et al.* [1]. Effective isolation is achieved using six piezoelectric actuators. In most circumstances, passive damping is preferable due to cost, complexity and reliability of active systems. The following section presents details about some of the common methods for passive damping.

III. Passive damping mechanisms

Passive damping is further separated into two classes: inherent and designed-in [2]. Inherent damping exists in a structure due to friction in mechanical joints, material damping properties, etc. Designed-in damping refers to damping that is added to the structure by design. This can be accomplished by integrating one or more of the following methods into the system.

a. Viscoelastic materials

Viscoelastic materials are elastomers with long chain molecules enabling them to convert mechanical energy to heat when subjected to strain. This energy dissipation can provide effective passive damping in structures. Viton is an elastomer that is chemically stable, and it is occasionally used in UHV systems. These materials are relatively incompressible (Poisson’s ratio ≅ 0.5), and when used as spacers between metal plate stacks, these materials are strained under compression. By the virtue of high compressive stiffness, they generate a resonant frequency typically between 10 Hz to 100 Hz. In addition, Viton has high internal damping and depending on the specific material, in the range of 0.05 to 0.3 times the critical damping [3].

Elastomeric materials for vibration isolation use hysteretic damping to dissipate energy. When these materials are deformed, internal friction causes high energy losses to occur. The loss factor, h, is used to quantify the level of hysteretic damping of a material. The loss factor is the ratio of energy dissipated from the system to the energy stored in the system for every oscillation. For viscous damping, the loss factor is generally expressed as a function of the damping coefficient, c, and critical damping coefficient, c_c, and is given by Eq. (1).

$$h = \frac{2c}{c_c} \tag{1}$$

A loss factor of 0.1 is considered as the minimum value for achieving significant damping. Loss factors for various materials are listed in Table 1.

Table 1: Approximate loss factor values for various materials [4]

Material	Loss Factor (η)
Aluminum	0.007 - 0.005
Steel	0.05 - 0.10
Neoprene®	0.1
Butyl Rubber®	0.4
ISODAMP® C-1002 thermoplastic	1.0

Oliva *et al.* [5] used a metal stack arrangement as shown in Figure 1 with Viton pieces (cross-sectional diameter ranging from 3.2 mm to 6 mm and length ranging from 5 mm to 60 mm) sandwiched between the plates to damp high frequency noise. A similar configuration of metal stacks with Viton is described by Binnig *et al.* [6] to isolate an STM from high frequency vibrations.

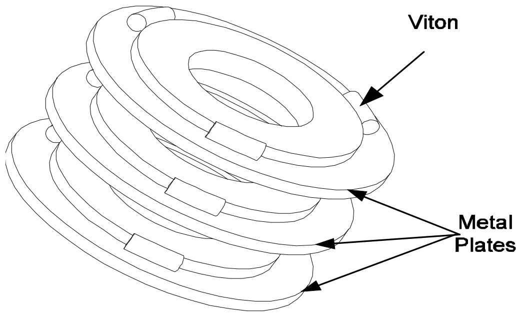

Figure 1: Stacks of plates with Viton [5]

Passive damping using viscoelastic materials is also accomplished with extensional damping treatments [7]. These treatments are broadly classified as unconstrained or free layer damping treatment and the constrained damping treatment. In the case of unconstrained damping treatment, a high modulus, high loss factor material is applied to a surface of the vibrating structure so that when the structure is subjected to cyclic bending, the damping material is subjected to tension-compression deformation. This configuration is illustrated in Figure 2. Constrained layer treatments are surface treatments where the damping material is sandwiched between the structure and a constraining layer. The constraining layer causes shearing in the damping material as the structure deforms. These types of damping treatments are commonly used to damp bending modes of the structure. The damping performance of such treatments is a function of the operating temperature, thickness of the viscoelastic material, initial structural damping, and bonding techniques. Although, most commercially available materials for these techniques are not UHV compatible, they may be applied to the exterior chamber (e.g. flange plates) or within sealed tubes, similar to the method described by Marsh *et al.* [8].

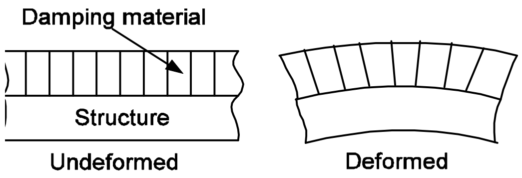

Figure 2: Unconstrained or free layer damping treatment

The two important limitations in using viscoelastic materials in UHV environments are temperature and outgassing rate. These materials cannot withstand temperature extremes; the ideal working temperature range for viscoelastic material is from 10 °C to 60 °C. The temperature limitation is optimized by selecting viscoelastic materials with different properties for coating. The second limitation is that the viscoelastic materials have relatively high outgassing rates. Outgassing rates for various materials used in UHV environment are listed in Table 2.

Table 2: Outgassing rates of various materials in UHV environment [9]

Material	Outgassing Rates (W/m^2)
Viton-A® (fluoroelastomers)	152×10^{-5}
Viton-A® (After vacuum baking)	2.7×10^{-7}
Buna	467×10^{-5}
Stainless Steel	1.2×10^{-8}

b. Magnetic devices

Magnetic eddy current damping is a viable solution for damping where high temperature extremes are a factor. Magnetic eddy current damping is a better alternative because of its vacuum compatibility and the ability to vary the damping coefficient. The principle of eddy current damping is shown in Figure 3. Electromagnetic forces are generated by the movement of a conductor through a stationary magnetic field or a time varying magnetic field through a stationary conductor, and can be used to suppress the vibrations of a structure [10]. These currents dissipate energy as they flow through the resistance of the conductor. The resulting force on the conductor is proportional to its velocity relative to the field and thus functions as a viscous damping element.

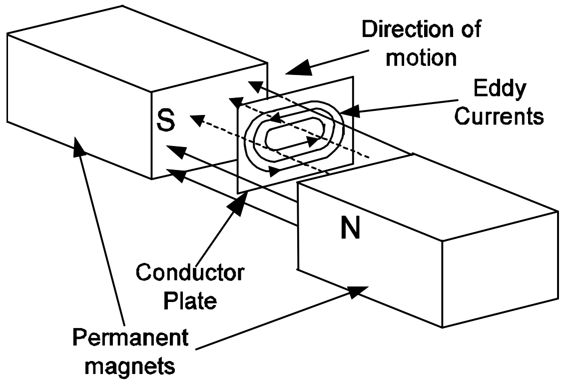

Figure 3: Principle of eddy current damping

Equations for calculating the damping coefficient for various types of configurations of magnets and conductors have been derived by several researchers. For example, the damping coefficient, c (Ns/m), for an eddy current damper as a function of the magnetic flux density, B, in Teslas, given the length of the conductor, L, in meters, cross-section area of the conductor, A, in square meters (m^2), and resistivity of the conductor material, r, in ohm-meters (Ωm) and is given by Eq (2) [11].K is a factor that accounts for losses from the damper configuration and other potential losses due to imperfections. It typically varies between 0.25 Ns/m and 0.35 Ns/m [12].

$$c = K\frac{B^2 hA}{r}\left[\frac{1}{2\left(1+\frac{L}{h}\right)}\right] \qquad (2)$$

c. Tuned mass dampers (TMD)

One other method of increasing damping of a structure is through the use of one or more tuned damping devices. Such a device could be in the form of a single degree of freedom system consisting of a mass on a linear spring. A tuned mass damper (TMD) is a damping device attached to the structure at or near an anti-node [2]. The principle of a tuned mass damper is that if a structure has relatively widely separated resonances, a tuned mass damper can be designed by ensuring that the damped natural frequency, w_D, is close to the frequency of the mode, w_1, to be damped [7]. The TMD splits the original single mode into two modes, as shown in Figure 4. The lower frequency peak corresponds to the damper mass and the structure moving in phase with each other, whereas the higher frequency peak corresponds to the mass and the structure moving out of phase. These kinds of dampers are most effective for single mode damping.

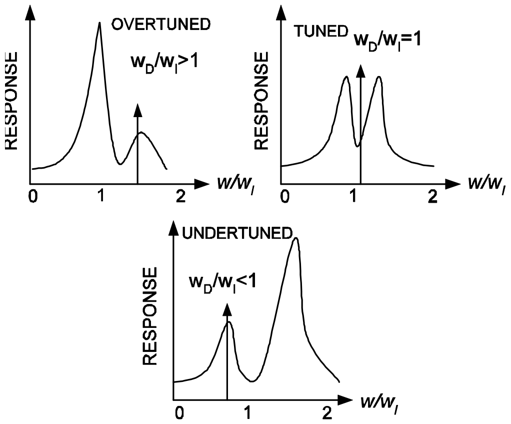

Figure 4: Effect of tuned mass damper on frequency response

d. Piezoelectric materials

Piezoelectric materials produce strain when subjected to an electric charge. This property is utilized in applications such as microactuators. A second property is that these materials produce a charge under strain. This property makes them well suited for sensing strain. In passive energy dissipation applications, the electrodes of the piezoelectric material are shunted with a passive electric circuit [13]. The electric circuit is designed to dissipate the electrical energy that has been converted from mechanical energy by the piezoelectric material. The shunting circuit generally consists of a resistor or resistor in series with an inductor. Shunting with a resistor and inductor, along with the inherent capacitance of the piezoelectric material, creates a resonant LRC circuit that is analogous to a tuned mass damper, except that it counters the vibration strain energy instead of the kinetic energy.

e. Shape memory alloys (SMA)

Some metallic alloys such as Nitinol (NiTi) exhibit the shape memory effect, which is suitable for generating force and displacement when the alloy changes phase during a heating and cooling cycle. Humbeeck *et al.* [14] investigated the use of shape memory alloys (SMA) for passive damping. The damping capacity is inherent in the material and is due to intrinsic defects of the martensite variants. The damping capacity of these materials can be optimized depending on the temperature, strain amplitude, frequency, alloy, and its thermomechanical treatment. Since strain amplitudes of 10^{-5} or higher can be achieved using these materials, it is suitable for applications where large amplitude vibrations (e.g. earthquakes and impact loading) must be damped.

IV. Low and high frequency vibration isolation in UHV systems

External vibrations that affect the performance of precision applications in UHV environment are divided into low and high frequency noises. This section discusses details such as the sources of these vibrations and some of the methods employed to minimize them. Figure 5 illustrates some of the different sources for the low and high frequency vibrations and their frequency ranges. Typical low frequency vibrations that disturb the system are between 10 Hz and 100 Hz. The common sources for these vibrations are people walking, vibrations excited by the equipment running at or near line frequency (60 Hz), floors and building walls undergoing shear and bending vibrations ($\approx$15 Hz to 25 Hz) [15].

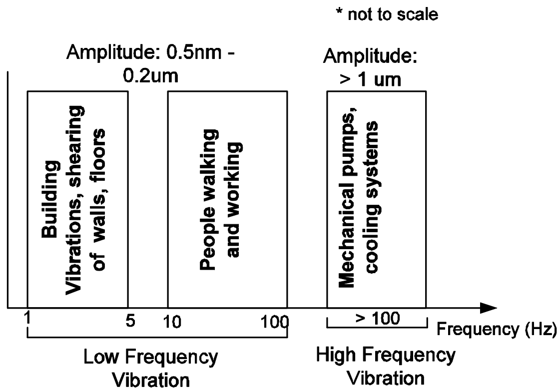

Figure 5: Sources for low and high frequency vibrations and their frequency range

Several methods are used to minimize the low frequency vibrations. A common practice is to mount the entire chamber on an optical table supported by vibration isolation legs. Alternatively, Drake *et al.* [16] isolated an STM from floor vibrations by suspending its platform by bungee cords from the ceiling. Chikkamaranahalli *et al.* [17] and Krapf *et al.* [18] have described a vibration isolation system (VIS) in which the entire vacuum system is mounted on a concrete slab supported by four air springs. In recent years for microscopes such as STM, operating under UHV condition, spring suspension is one of the most popular ways to minimize low frequency vibration. The eigen frequency, *f*, of a linear one stage spring suspended system is inversely proportional to the square root of the static elongation of the spring, Δz, in meters (m) and is given by Eq.(3)

$$f = \frac{1}{2p}\sqrt{\frac{g}{\Delta z}} \qquad (3)$$

The advantage of using springs in a UHV environment is that it allows high bake-out temperatures [19]. However, metal springs have the disadvantage of low internal damping and hence additional damping is necessary to reduce the excessive motion amplitudes at resonance [20]. The second disadvantage of the spring suspension is the lack of a well-defined position of the microscope with respect to the surrounding vacuum chamber. This deficiency becomes a problem when using mechanical manipulation systems, *in situ* sample transfer, and tip exchange. All these tasks demand spatially well-defined locations and the ability to exert forces on the microscope. In addition, metal springs with low spring constants yield low resonant frequencies with high quality factors. Hence, additional damping elements are necessary to reduce amplitudes at resonance. Prior to using the spring suspension, Binnig *et al.* [21], used superconducting magnetic levitation at liquid helium temperature to isolate an STM from external vibrations. Most of the recent UHV-STM and AFM use magnetic eddy current damping for low frequency vibration isolation [22] because of the advantages described in the previous section.

Vibrations at frequencies greater than 100 Hz can be considered as high frequency noise vibrations. The common sources for these vibrations are the mechanical pumps and cooling systems that run the vacuum system and the instrument. One of the ways to minimize the effect of these vibrations is to house the microscope inside an acoustical chamber.

V. Design case studies

In this section, we discuss the implementation of some of the previously discussed passive damping methods for vibration isolation for scanning tunneling microscopes. One such system is described and analyzed by Okano *et al.* [20].

The system consists of two stages of coil spring suspension isolation. Each end of the coil springs is terminated by a rubber ring to attenuate the high frequency acoustic vibration from propagating through the solid frames, columns, and coil springs. Additional damping is provided in the form of eddy current dampers that are attached to the lower part of the structure. Experiments were conducted to study the isolation characteristics with and without the eddy current dampers. With a light push on the structure, the oscillations continued for about 30 minutes without the dampers. This implies in the absence of dampers, the vibration transmission drops off very steeply in the high frequency range, but the tunneling assembly will vibrate its eigen frequency for a long time once the isolator has been perturbed. With the eddy current dampers attached, some of the vibrations with high peaks were eliminated and the oscillation attenuated within a few seconds. A metal stack isolator was also used to suppress some of the high frequency vibrations. The schematic of the metal stack arrangement including the spring stages is illustrated in Figure 6. Three pieces of Viton rubber are used between each metal plate.

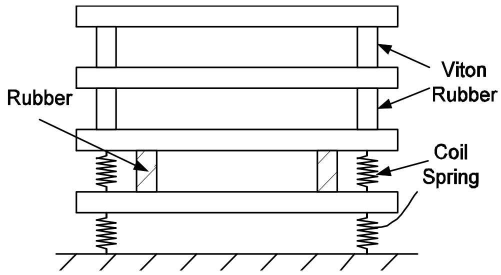

Figure 6: Five-fold metal stack with Viton rubber, Okano *et al.* [20]

Both the isolation systems were tested. The transfer function was obtained by placing the assembly on a mechanical shaker and measuring the acceleration magnitudes at the base and the sample holder at the same time as a function of vibration frequency. Isolation of -60 dB at 22 Hz and 320 Hz was obtained for the coil spring suspension and the five-fold metal stack isolator, respectively. The performance of the metal stack isolator was enhanced by replacing rubber with small coil springs.

Krapf *et al.* [18] used an alternative approach rather than the conventional spring suspension. This arrangement is shown in Figure 7. The microscope assembly is mounted on a UHV flange that is linked to the chamber through a welded bellows. The supporting flange of the microscope is on a rigid support, which consists of an iron tripod mounted rigidly on a concrete block. The flexible bellows, which links the microscope's supporting flange to the UHV-STM chamber, is intended to reduce the vibrations transmitted from the surrounding system. The advantage of using such a VIS over the spring suspension system is that the effects of building vibrations on the STM are minimized to a large extent. This can also be achieved using a longer spring suspension but is not feasible in a compact vacuum chamber. Dynamic testing of the system indicated that a gain value of -80 dB was obtained for frequencies less than 50 Hz.

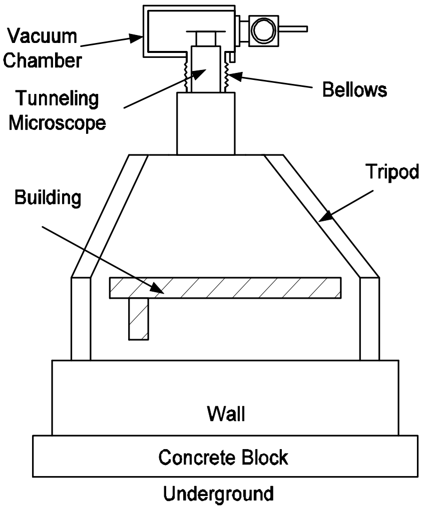

Figure 7: Overall design of the vibration isolation system, Krapf *et al.* [18]

VI. NIST UHV-STM

We are currently building and testing a UHV-STM at NIST. The complete design and the working principle of the STM are explained in detail by Gilsinn *et al.* [23]. The schematic diagram of the vibration isolation system is illustrated in Figure 8.

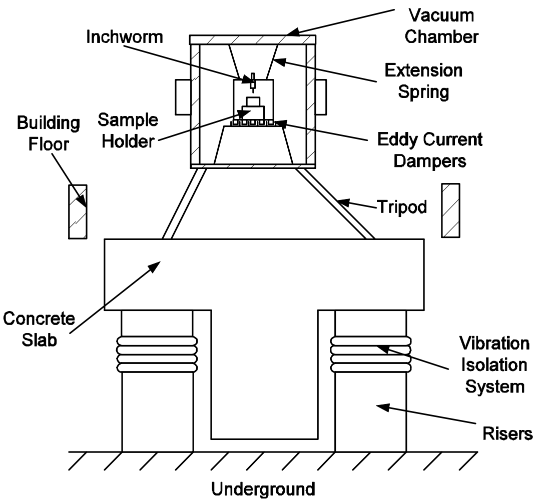

Figure 8: Vibration isolation system for the UHV-STM at NIST

The vacuum chamber is mounted on a commercially available shear damped tripod that is bolted to the concrete slab. The concrete slab is supported on four pneumatic vibration isolation legs. The vibration displacement magnitude, *a*, is then computed as a function of the acceleration magnitude, *M*, and the oscillating frequency, *f*, using Eq (4).

$$a = M \Big/ (2\pi f)^2 \qquad (4)$$

The magnitude spectra of the acceleration measured on the laboratory floor and on the STM frame is shown in Figure 9.

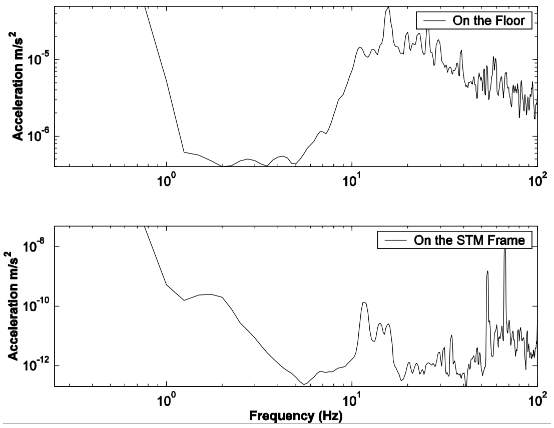

Figure 9: Acceleration Magnitude on the Lab Floor and STM Frame

The maximum acceleration magnitude on the laboratory floor is 4.90×10^{-4} m/s^2 [i] at 16 Hz[ii], corresponding to an amplitude of 50 nm. The acceleration magnitude as measured on the instrument's supporting frame is only 2.45×10^{-7} m/s^2 [i], at 68 Hz [ii], which corresponds to a displacement amplitude of approximately 71 pm. This is still a significant amount of excitation to the STM, and further isolation and damping is necessary to reduce this amplitude to only a few picometers. Spring suspension and magnetic eddy current damping is therefore used to further enhance the isolation characteristics of the STM. The details of the eddy current damping are explained in detail by Chikkamaranahalli *et al.* [17]. The displacement amplitude spectra for the system with and without the eddy current dampers were measured. It was clearly observed that the oscillation attenuated within a few seconds with the eddy current damper as compared to a few hours without the damper.

The STM with the current damping setup was tested in air for atomic scale resolution. Highly oriented pyrolytic graphite (HOPG) is imaged in a constant height mode with a scan rate of 2.54 Hz, bias voltage of 100 mV, and a current set point of 300 pA. The result of a 5 nm × 5 nm scan area is shown in Figure 10.

[i] *Uncertainty in amplitude measurement is* $\pm 4.9\times10^{-6}$ *m/s*^{2}*, k = 2*
[ii] *Uncertainty in frequency measurement is ± 0.25 Hz, k = 2*

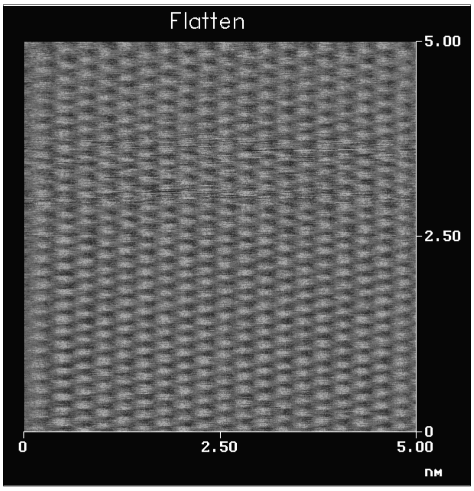

Figure 10: Topographical image of highly oriented pyrolytic graphite (HOPG) in air showing the atomic corrugation (Scan time: ~60 seconds)

VII. Conclusions and future work

Damping is one of the important design factors to be taken into consideration for precision applications. To achieve damping, two conditions must be met: significant strain energy must be directed into high loss mechanisms for all modes of interest and the energy in the mechanism must be dissipated. This article discussed some of the passive damping methods that are currently used in UHV environments and proposed a few other damping methods that are compatible in UHV environments. The implementations of some of these methods in damping devices such as an STM have been reviewed. The damping procedure used in isolating the UHV-STM that is being built and tested at NIST from external vibrations has been discussed. The UHV-STM with magnetic eddy current dampers is successfully used to scan highly-oriented pyrolytic graphite (HOPG) in air.

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